ESL-TR-93/10-05

LoanSTAR Monitoring and Analysis Program

Potential Operation and Maintenance (O&M) Savings in the Clinical Science Building at UTMB

Submitted to the
State Energy Conservation Office of Texas
By the
Monitoring Analysis Task E

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DISCLAIMER

EXECUTIVE SUMMARY

This report presents the results of a study of the potential energy savings due to optimizing the Heating, Ventilation and Air Conditioning (HVAC) operation schedule in the Clinical Science Building at University of Texas Medical Brach (UTMB) Galveston, Texas. An optimized HVAC operation schedule has been developed using the simplified model analysis with the LoanSTAR measured hourly data and the EMCS measured operation parameters at UTMB. An annual savings of $73,700 can be realized by implementing this optimized schedule by changing the EMCS control program. The majority of the energy savings are due to the reduction in chilled water consumption and the substantial reduction of reheat. Our analysis indicates that the indoor comfort level will not be degraded by this measure. It can reduce the building's current annual energy costs by $73,700 or 21%.

This report discusses a simplified model analysis and the methodology of identifying one type of O&M improvement and summarizes the potential savings from this measure.

Table of Contents

POTENTIAL OPERATION AND MAINTENANCE SAVINGS IN THE CLINICAL SCIENCE BUILDING AT UTMB

1. INTRODUCTION

The Clinical Science Building is a 124,870 ft^2 six story facility connected directly to the west side of John Sealy Hospital. The exterior surface is made of brick and is approximately 44,800 ft^2. Clinical Science building is a major teaching building on the UTMB campus. Although teaching and laboratory work are conducted on a 5-day basis, the building also houses a morgue facility, which is open 24 hours. Other selected areas (allergy labs, for example) are also used at odd hours. This building was placed in service on January, 1970 and is expected to serve UTMB for many more years.

The building is provided 45% outside air by one 200 hp constant volume dual duct AHU, capable of supplying 126,000 cfm. Currently the fan is supplying air at a rate of 1 cfm/ft^2. Chilled water and steam is supplied by the main chiller plant. Steam is converted into hot water by a hot water converter (1,7500 lb./hr). A variable frequency drive chilled water pump (50 hp, 2,050 gpm) supplies chilled water to the AHU. There is one other small AHU (7.5 hp) supplying 8,500 cfm of air to the 3rd floor lecture room. The building HVAC system is operated 24 hours a day all year long. Lighting in the building is provided exclusively by fluorescent fixtures. Lighting intensity varies widely throughout the building.

The hourly building energy consumption data (electricity, chilled water & Steam) are being measured by the LoanSTAR program as well as by Steffa Energy Management & Control System (EMCS) at this building (see Appendix B for detail). According to the LoanSTAR measured results, this building consumed 4.33 million kWh in electricity, 22,975 MMBtu chilled water and 13,377 MMBtu steam from September 1, 1992 to August 30, 1993. The total cost of these utilities comes out to be $350,500/yr. or $2.80/$ft^2$. The following unit price has been used to calculate the total utility cost: $0.02659/kWh, $7.30/MMBtu for chilled water and $5.055/MMBtu for steam. Figure 1 and Table 1 show the breakdown of energy consumption and cost.

Table 1: Summary of the Annual Energy Consumption and Cost at Clinical Science Building
September 1, 1992 - August 30, 1993

	Electricity million kWh	Chilled Water MMBtu	Steam MMBtu	Total Cost
Consumption	4.33	22,975	13,394	
Costs ($)	$115,200	$167,700	$67,600	**$350,500**
% of Total Cost	33%	48%	19%	

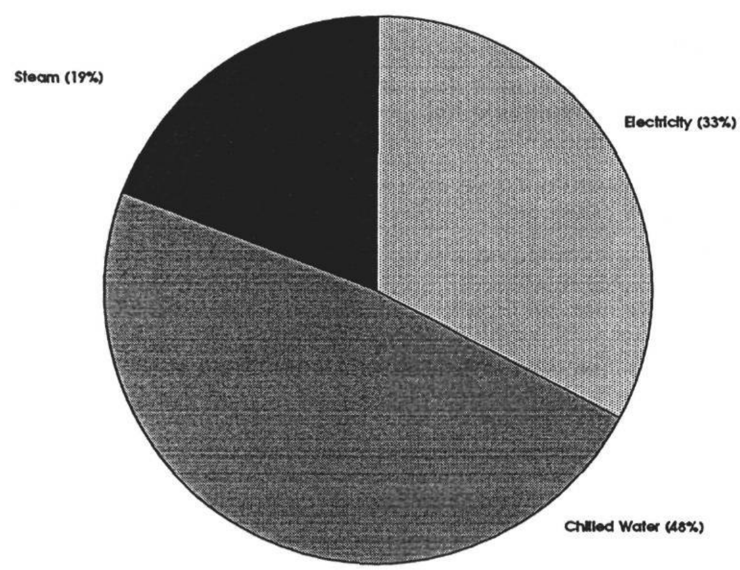

Figure 1: Energy Cost Distribution for the Clinical Science Building; Total Current Annual Energy Cost is $350,500

Figure 2 shows measured average daily chilled water and steam energy consumption vs. ambient temperature. Substantial amount of steam is used on very hot summer days, and the consumption increases with the temperature drop, indicating that substantial reheat is present and also reflecting a large amount of domestic hot water consumption.

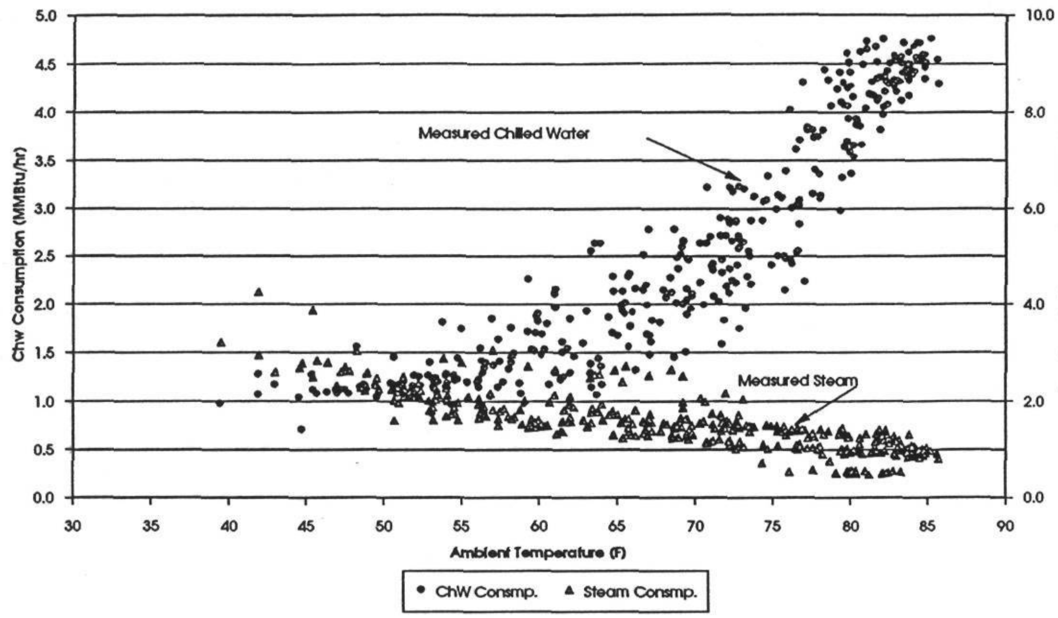

Figure 2: Measured Chilled Water and Steam Energy Consumption vs. Average Daily Ambient Temperature. Data were measured from August 1, 1992 to July 31, 1993.

Both the air handling units and their associated equipment are controlled by the newly installed Steffa Energy Management Control System. It can continuously monitors and control different parameters like cold deck temperature, hot deck temperature & speed of the fans based on space and ambient temperatures.

This report describes a study of potential O&M improvements conducted for the Clinical Science Building at UTMB. It briefly describes the methodology used to identify different O&M measures, presents simplified HVAC system model used for the present O&M analysis and HVAC operation optimization. The selected O&M measure is discussed in detail with some recommendations and conclusions

2. METHODOLOGY

The methodology used to explore the O&M opportunities is outlined below:

1. **<u>LoanSTAR information base browse:</u>** the LoanSTAR information base includes:
   (i) the LoanSTAR Data Base (LSDB), which contains continuously measured hourly energy and weather data;
   (ii) the Site Description Notebook, which contains updated information about the building's Heating, Ventilation and Air Conditioning (HVAC) system, lighting, building envelope, occupancy and other relevant information from the audit report;
   (iii) weekly inspection plots (IPNs), which give an updated performance of the building every week;
   (iv) the Monthly Energy Consumption Report (MECR), which details monthly energy performance ;
   (v) the Annual Energy Consumption Report (AECR), which summarizes yearly energy performance and the overall energy performance history of the building.

   Browsing this information base gives O&M staff a draft list of O&M candidates in the building.

2. **<u>Site visit/system examination:</u>** The purpose of the site visit includes:
   (i) discussing potential O&M measures with UTMB physical plant personnel;
   (ii) verifying information gathered from LoanSTAR database by a simple walk-through with the building operator;
   (iii) examining the possibility/feasibility of potential O&M measures;
   (iv) exploring new O&M measures; and
   (v) collecting system information, such as cold deck and hot deck temperature schedules, air flow, and nighttime setback schedule as well as miscellaneous information from the EMCS such as EMCS measured energy performance.

3. **<u>Data quality check:</u>** before using the LoanSTAR data to estimate O&M savings, they are compared with EMCS measured data. If the two sets of data are fairly consistent,

the LoanSTAR data is used in the analysis without correction. If the LoanSTAR measured data and EMCS measured data are seemed unacceptably different, the LoanSTAR data is checked using other methods. This quality check provides reliable data for the savings analysis. The data quality check in this building indicates that the LoanSTAR measured data are reliable (see Appendix B).

4. **System modeling and calibration:** The HVAC systems and the building are modeled by a set of equations, and programmed into a computer code. The simplified computer model uses measured daily average ambient and dew point temperatures to predict daily average hourly chilled water and hot water energy consumption. Finally, the predicted energy consumption is compared with measured consumption. If the predicted consumption matches measured energy consumption, then the simplified computer model and its associated parameters, such as air flow, cold deck and hot deck settings, and internal gains are calibrated. Otherwise, calibration is required which involves adjusting parameter estimates such that better agreement with monitored data is achieved.

5. **O&M simulation & savings calculations:** The cold deck and hot deck schedules are optimized to consume minimum energy while the following conditions are satisfied:

i) room temperature should be unchanged;
ii) room relative humidity should be less than 60%;
iii) the air flow rate to each room should not change;
iv) the maximum CFM through the cold and hot decks and ducts should be less than their capacities or design values; and
v) there should be no extra implementation costs involved.

Energy savings are taken as the difference between the base model (calibrated model) predicted annual energy consumption and optimized model predicted annual energy consumption.

6. **Feedback from UTMB physical plant personnel:** UTMB personnel comments on the proposed optimized schedule and necessary information to modify the proposed schedule if needed is provided. The simplified model simulation may indicate that some of the EMCS measured values are incorrect. These parameters are discussed during the feedback meeting and are jointly measured by both LoanSTAR and UTMB personnel.

7. <u>Refinement of simulation & savings calculations:</u> All the suggestions and findings are incorporated into the simplified model and potential savings are recalculated.

8. <u>Short-term test of optimized schedule and implementation.</u> The fixed temperature setting for the cold deck and hot deck are derived from the optimized schedule under certain ambient temperature conditions. UTMB staff disable the EMCS system temporally and use the suggested settings instead for a few days. Although this test would not show the full potential of optimized schedule savings, it provides opportunities to expose some hidden problems, if any. If there are no problems after this test, the optimized schedule is programmed into the EMCS system by the UTMB staff.

3. SIMPLIFIED MODEL & ITS CALIBRATION

3.1. Simplified Model and Input Data

The schematic of the air handling unit is shown in Figure 3. The air handling unit has a air supply capacity of 126,000 cfm, with a total outdoor air intake of 56,700 cfm. According to the EMCS data, the following parameters were recorded on July 15, 1993 at 7:54 pm.

Cold deck temperature:	54.5 ^{o}F
Return air temperature:	75 ^{o}F
Avg. space temperature:	72 ^{o}F
100% outside air preheat temperature:	84 ^{o}F
Pretreat preheat temperature:	83.4 ^{o}F
Pretreat supply temperature:	54.1^{o}F
Hot deck temperature:	81.3 ^{o}F

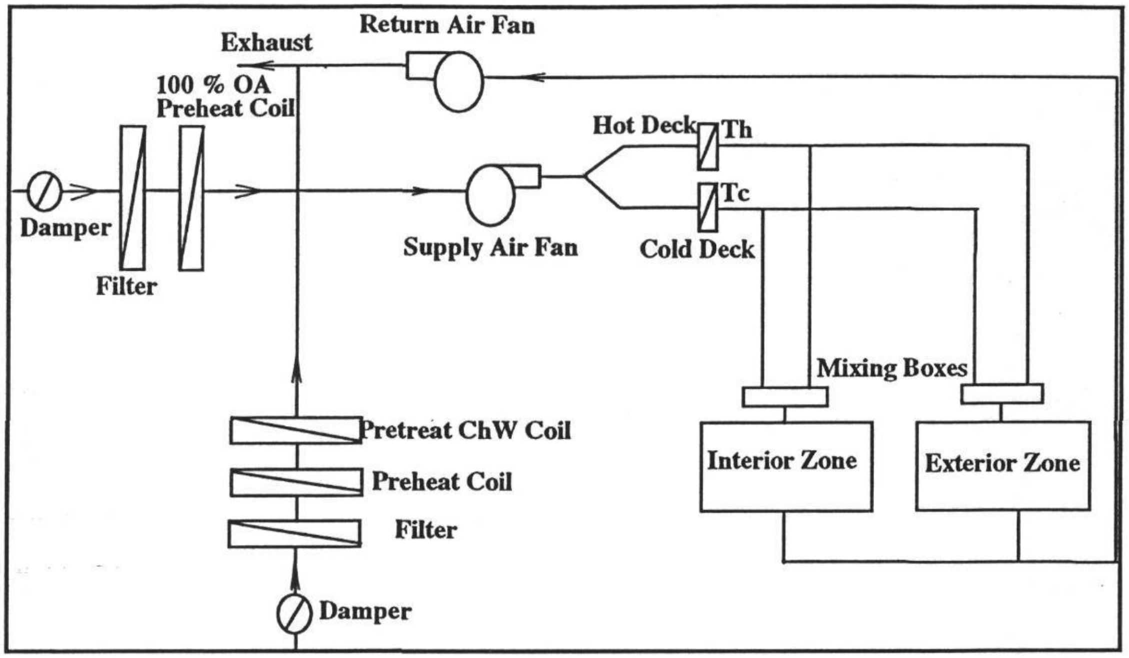

Figure 3: Schematic of Air Handling Unit

The main equations of the simplified model are presented in Appendix A. The basic parameters used in the model are discussed below.

According to the information supplied by the UTMB staff, the conditioned area is approximately 87,400 ft^2. The building has been divided into two zones: an interior zone and an exterior zone. The exterior zone is taken as the sum of areas which are directly connected with the exterior envelopes. According to the building floor plans (Figure 4), interior zone covers 52,450 ft^2 and exterior zone 34,950 ft^2. The internal heat gain is taken as 3 W/ft^2 based on the measured lighting load capacity, while a factor of 0.8 is used to account for the heat gain reduction at night. The number of people are calculated by assuming one person for every 120 ft^2 of conditioned area, and the sensible and latent loads due to people are calculated by assuming standard losses by normal office workers [3]. The domestic hot water consumption is estimated to be 0.9 MMBtu/hr. Figure 4 shows a typical floor layout of Clinical Science Building.

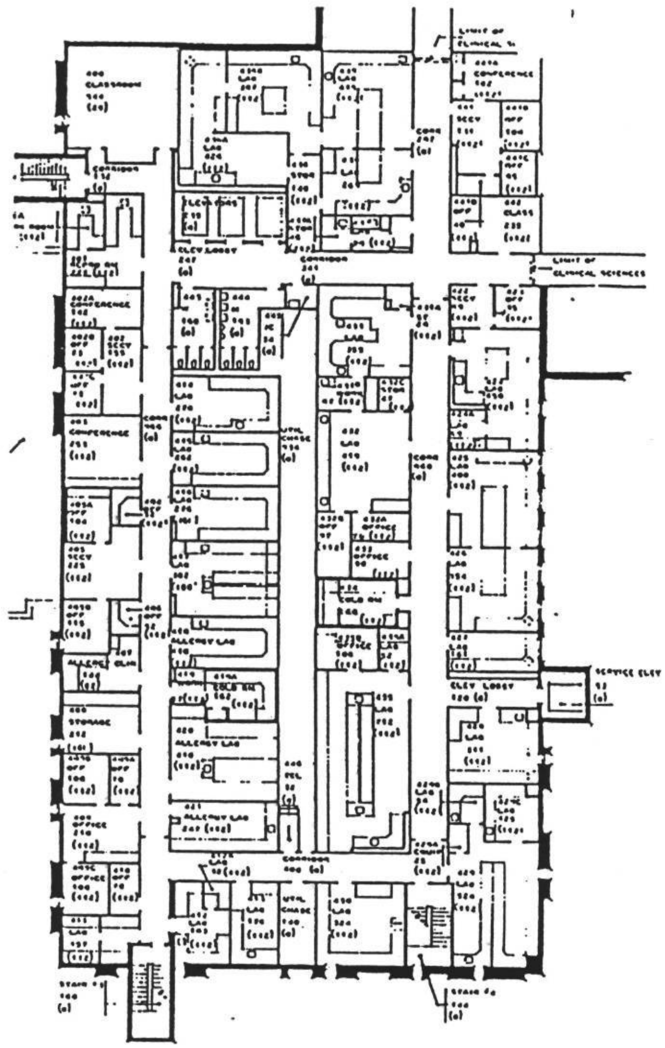

Figure 4: Typical Floor Plan in the Clinical Science Building

The building envelope area is calculated to be 44,800 ft^2, which includes 1,730 ft^2 of window area. Heat transfer coefficient value of 0.1 Btu/ft^2-^oF-hr was assumed for walls and 1 Btu/ft^2-^oF-hr for windows.

The air infiltration rate is taken as 0.4 ach (air change number of building volume in one hour) for the exterior zone and 0.2 ach for the interior zone. The interior zone receives infiltration through exterior doors and corridors

3.2. Model Calibration

The chilled water and steam energy consumption were predicted with the simplified model using measured daily average temperature from August 1, 1992 to July 31, 1993. Figure 5 permits a visual comparison of the measured energy consumption with model simulated energy consumption. The horizontal axis is ambient temperature while vertical axis is daily average chilled water and steam energy consumption. It shows that the simulated data fits well with the measured data. The predicted daily average chilled water consumption was 0.5% lower than measured values while the predicted steam consumption was 1% higher than the measured values over a period from August 1, 1992 to July 31, 1993. The standard mean square root errors of predictions are 0.42 MMBtu/hr and 0.13 MMBtu/hr for chilled water and Steam, respectively. The coefficient of variation are 15% and 7% for chilled water and steam, respectively.

The LoanSTAR measured chilled water energy consumption data is also compared with EMCS measured data on hourly basis for 24 hours. The comparison results shows that LoanSTAR measured steam consumption is with in 4% of measured data by EMCS (please see Appendix B for details).

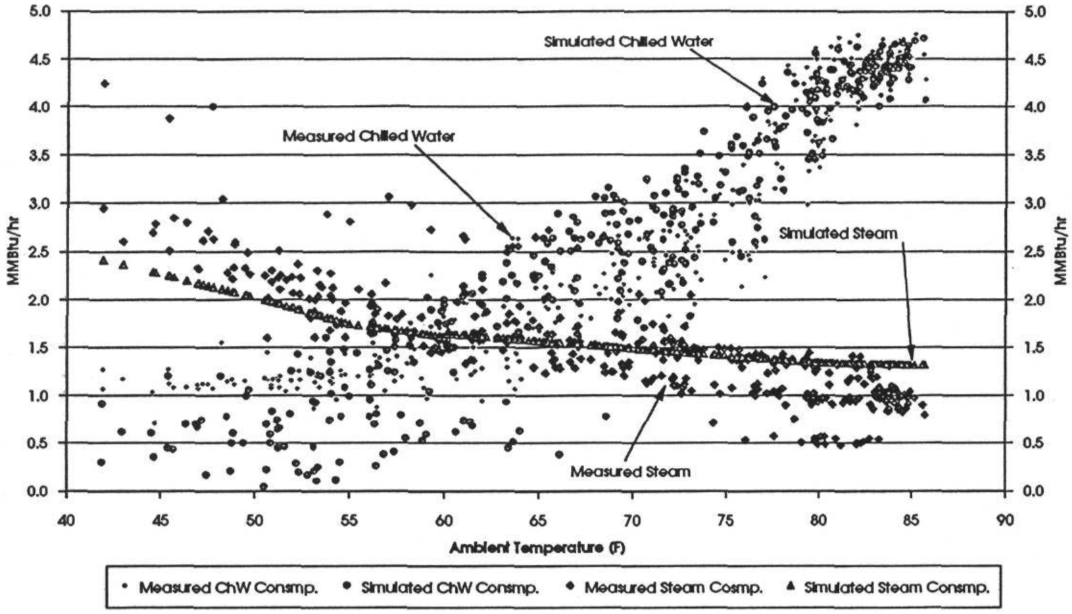

Figure 5: Comparison of Simulated and Measured Average Daily Energy Consumption (August 1, 1992 to July 31, 1993)

Figure 6 shows the comparison between simulated and measured energy consumption when plotted against time. It shows that the simplified model matches very well with the daily variation. The model predicted steam consumption, however is higher than measured values from August 1992 to October 1992 and from June 1993 to July 1993. This difference could not be explained. One possible reason could be some operational/schedule changes. The model predicted chilled water consumption, however is lower than measured values from November to April. This difference may be due to the fact that the model does not take into account the increase in chilled water consumption when preheat coils are used (i.e. when ambient temperature is less than 55 ^{0}F) and the chilled water coils are hit with air after being preheated.

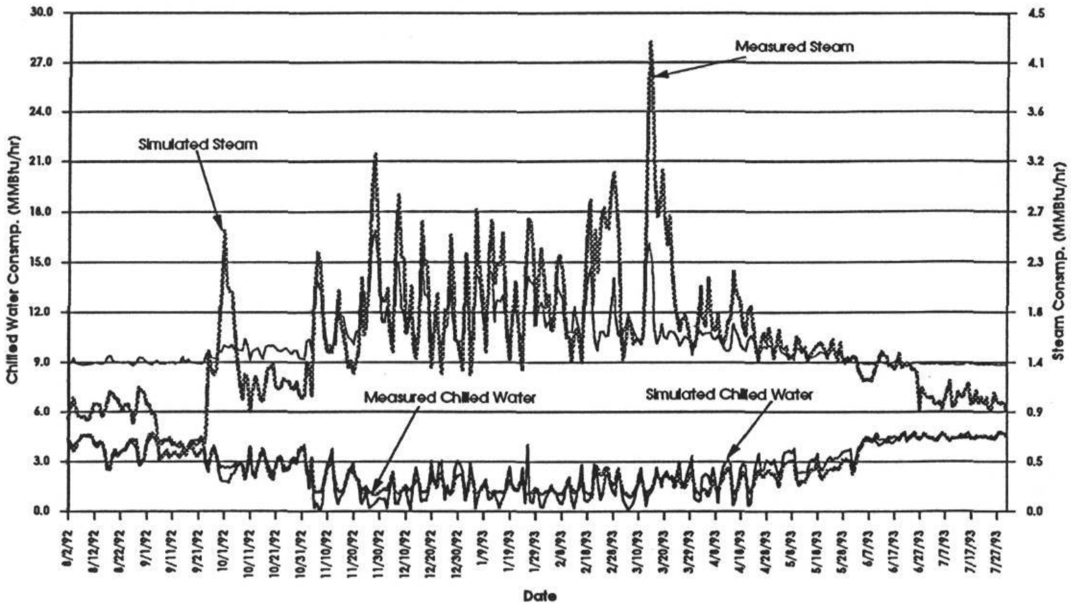

Figure 6: Comparison of Simulated & Measured Daily Average Energy Consumption (August 1, 1992 to July 31, 1993)

The calibrated simplified model was used calculate annual energy consumption using bin data for outdoor temperature. Due to the lack of measured hourly dry bulb and dew point temperatures for Galveston, measured hourly data from July 1, 1992 to June 30, 1993 for Houston was used to generate bin temperatures. Figure 7 shows the number of hours in each bin. The horizontal axis is the bin temperature, where 24 bins are used with a spread of 3 °F in each bin. The number of hours under a certain temperature during a full year are shown on the vertical axis. From the graph we can see that most of the hours are between 50 °F and 90 °F .

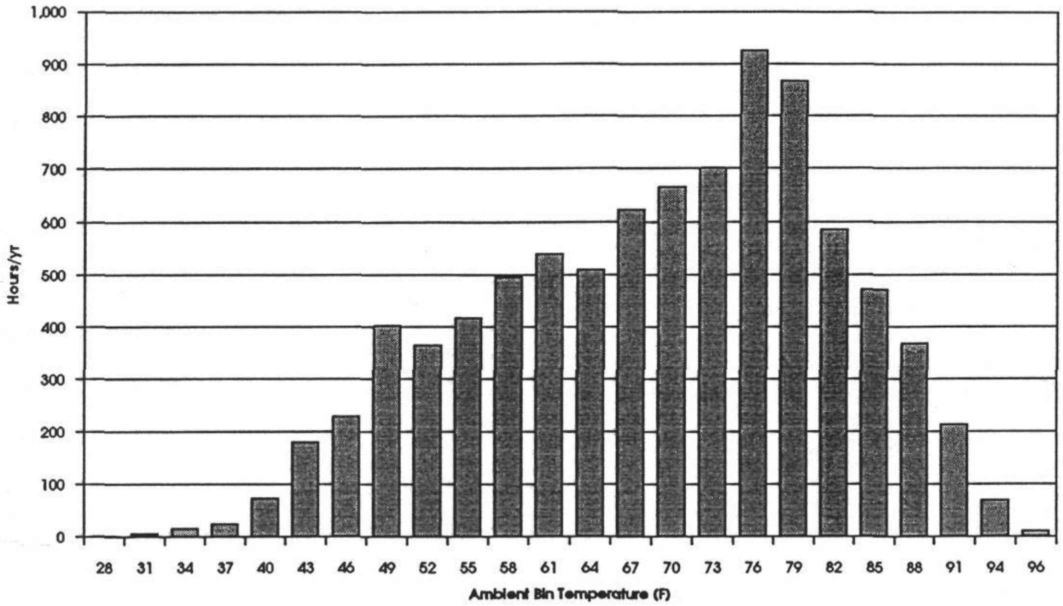

Figure 7: Houston Bin Temperature Chart, Generated using LoanSTAR Measured Hourly Temperature from July 1, 1992 to June 30, 1993

The mean coincident dew point temperatures are plotted as a function of the ambient dry bulb bin temperature in Figure 8. The figure shows that the dew point increases with the ambient temperature when the ambient temperature is lower than 80°F and then remain more or less constant when the ambient temperature is higher than 80°F. The fixed dew point temperature indicates that the absolute moisture content does not change when the ambient temperature is higher than 80°F. Consequently, the sensible load increases with temperature while the latent load does not change when the ambient temperature is higher than 80°F.

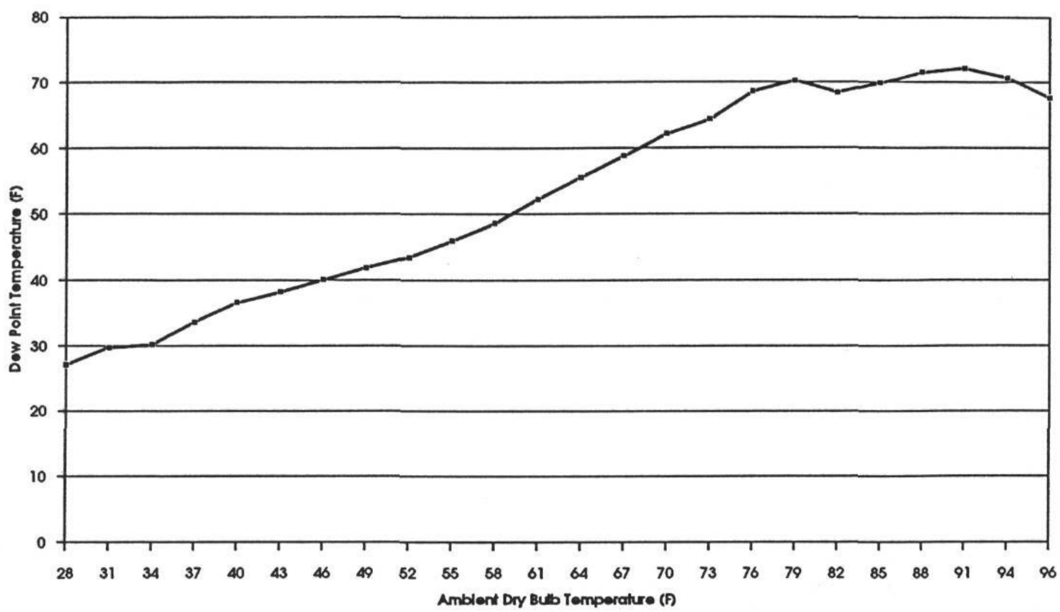

Figure 8: Mean Coincident Dew Point Temperature as a function of Ambient Dry Bulb Temperature for Houston from July 1, 1992 to June 30, 1993.

The comparison of measured and predicted annual energy consumption is given in Table 2. It shows that the calibrated model has a high level of accuracy in predicting annual energy consumption.

Table 2: Comparison of Measured & Simulated Energy Consumption (August 1, 1992 to July 31, 1993)

Item	Chilled Water	Steam
Measured Consumption	22,975 MMBtu	13,378 MMBtu
Simulated Consumption	22,962 MMBtu	13,564 MMBtu
Difference	13 MMBtu	186 MMBtu

Table 3 summarizes values of key parameters used in the calibrated simplified model and in the baseline setting of the EMC system. The fraction of return air is slightly different between the calibrated model and the value calculated from design parameters.

This value is adjusted to match chilled water consumption sensitivity to ambient temperature.

Table 3: Summary of the Model Calibration Parameter Adjustments.

Item	Schedule (EMCS)	Schedule (Model)
Air flow (cfm)	126,000 (Blue prints)	107,100
Outside air fraction	0.45 (Blue prints)	0.40
Main-cold deck temp. (^{O}F)	54.5	54.5
Pretreat chilled water coil temperature (^{O}F)	57.0	57.0
Hot deck (^{O}F)	If T0<80 then Min(100, 80+0.5*(80-T0)) Else 80	If T0<80 then Min(100, 80+0.5*(80-T0)) Else 85
Preheat coil temperature (^{O}F)	If T0<46 then 51 Else T0	If T0<55 then 55 Else T0
Return air temperature (^{O}F)	77	77
Room air temperature (^{O}F)	72	72

4. OPTIMIZE MAIN & PRETREAT COLD DECK SCHEDULES

The goal of optimizing the main cold deck, pretreat cold deck and main hot deck by adjusting their temperature schedules is to minimize the energy consumption while maintaining all comfort levels and also avoiding costly retrofit measures. In order to maintain indoor comfort levels, the following conditions should be satisfied: 1) the main cold deck supply temperature should not be greater than 62 ^{O}F during cold winter days, and should be low enough to maintain room comfort during hot summer days; 2) the pretreat cold deck supply temperature should not be greater than 57 ^{O}F during the cooling mode (T0>60); 3) the hot deck supply temperature should not be lower than 80 ^{O}F during hot summer days; and 4) the room relative humidity should be within the range of 25% to

60%. In order to avoid retrofit costs, the following constraints exist: 1) no reduction in air flow is allowed; 2) air flow rates through hot and cold ducts should not exceed design limits; and 3) no frequent manual operations should be involved.

The optimization process is an iteration process currently. A best operation schedule is chosen first based on O&M staff knowledge. Then, energy (chilled water and steam) and the mechanical operation performance (air flow through cold and hot ducts) are predicted using the simplified model. After energy and mechanical performance is compared with the best operation schedule known so far, modifications are made and a new simulation is performed. This process is repeated until the operation schedule is considered the best possible.

Table 4 lists the base and the optimized operation schedules for main cold deck, main hot deck and pretreat cold deck. Main hot deck schedule is changed to satisfy the condition that air flow rate through each duct component should not exceed design limits. The base and the optimized schedules are also shown in Figure 9.

Table 4: Comparison of Operation Schedules

Item	Base	Optimized

Main cold deck	54.5 °F	If T0>60 then
		Min(62, 62-0.125*(T0-60))
		Else
		62

Pretreat cold deck	57 °F	If T0>60 then
		Min(57, 56-0.05*(T0-60))
		Else
		57

Main hot deck	If T0<80 then
	Min(100, 80+0.5*(80-T0))
	Else
	85
		If T0<80 then
		Min(85, 85-0.25*(T0-60))
		Else
		off

Figure 9 shows the base and optimized schedule for main cold deck, main hot deck and pretreat cold deck air temperatures. Obviously, this optimized schedule can reduce chilled water and Steam consumption substantially.

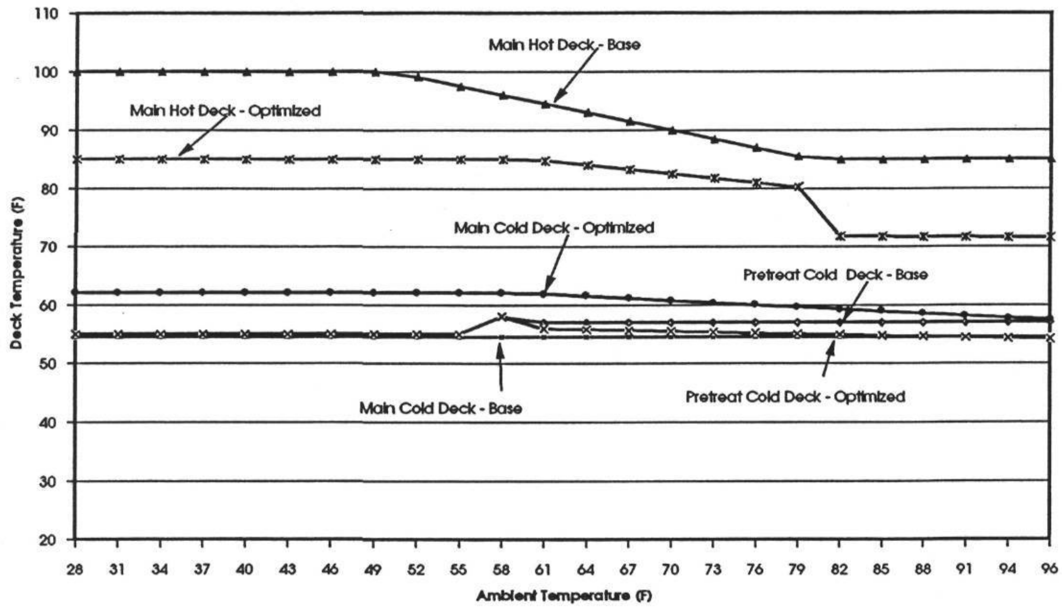

Figure 9: Base and Optimized Deck Schedules

The optimized schedule changes all the decks smoothly with ambient temperature, which can be performed by EMCS without any major changes or effort.

The energy and the mechanical performance under an optimized operation schedule are compared with the base performance in the next section.

5. SIMULATED RESULTS AND DISCUSSIONS

5.1 Thermal Energy Saving Potential:

The calibrated simplified model has been used to calculate the chilled water consumption, Steam consumption, room relative humidity, and air flow rate through cold and hot ducts at each bin temperature and its coincident dew point for both the base and optimized schedules. The annual energy consumption is calculated by summing the products of the energy consumption and number of hours at each bin temperature over all bin temperatures.

Figure 10 compares the optimized energy performance with the base energy performance. The horizontal axis is the ambient bin temperature. The vertical axes are the energy consumption for chilled water and the steam in MMBtu/hr. It shows that the optimized schedule can reduce chilled water consumption by 1 MMBtu/hr and steam consumption by 0.4 MMBtu/hr regardless of the ambient temperature. The simultaneous reductions of the chilled water and the steam consumption indicate that the major part of the savings are due to elimination of simultaneous cooling & heating. The relative larger chilled water savings indicate that the optimized schedule will remove less moisture, which can cause a higher room relative humidity.

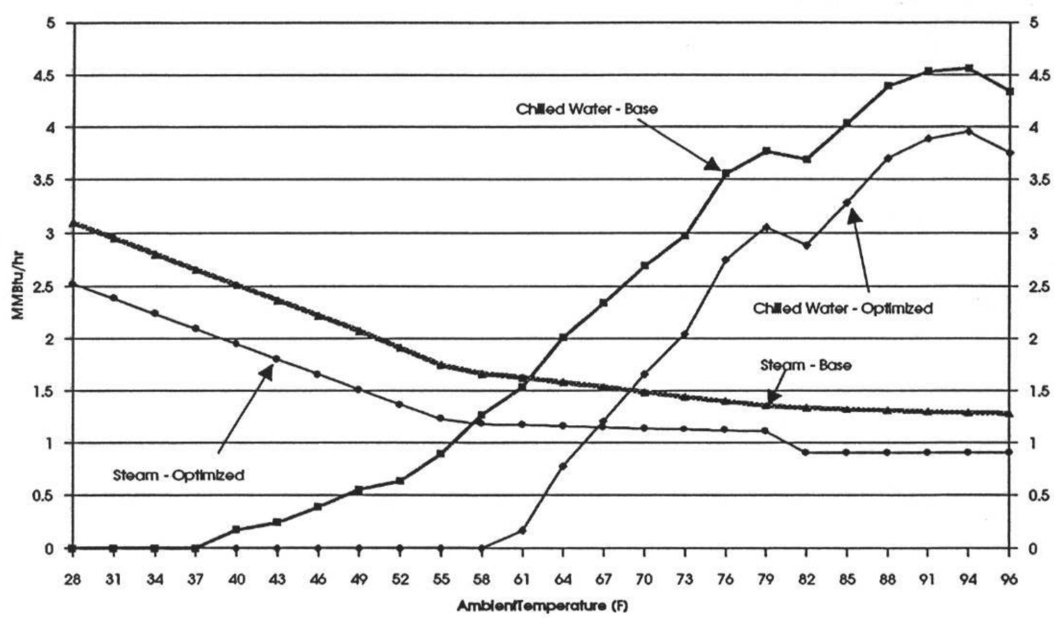

Figure 10: Comparison of the Predicted Chilled Water and Steam Energy Consumption Under Base and the Optimized Operation Schedule

Figure 11 compares the predicted room relative humidity levels under the optimized and the base schedule. The predicted room relative humidity under the base schedule was consistent with the EMCS measured values. The optimized schedule can increase the room relative humidity to 56%, which is about 5% higher than the base schedule value. Recent studies [4] have found that room relative humidity levels have less impact on comfort levels than was thought earlier and there is now a tendency to enlarge the relative humidity comfort zone from 25% ~ 60% to 25% ~ 70%.

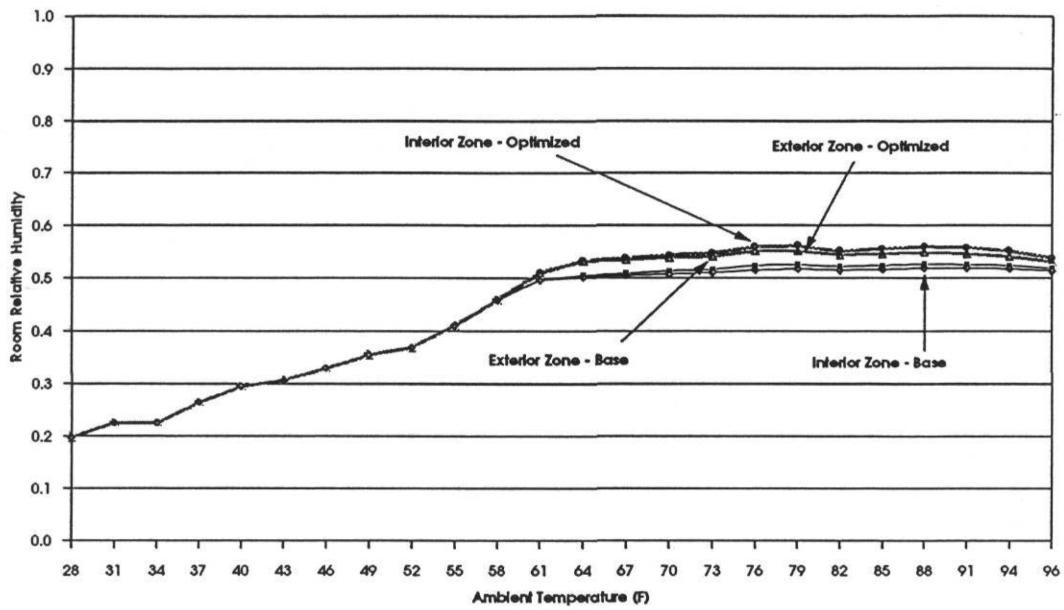

Figure 11: Comparison of the Predicted Room Relative Humidity under the Base and Optimized Operation Schedules

Figure 12 compares the predicted air flow rates through cold and hot air ducts under both the base and the optimized schedules. The base schedule has a cold air flow range of 69,200 cfm to 78,600 cfm and a hot air flow range of 28,500 to 37,900 cfm, while the optimized schedule has a cold air flow range of 67,800 cfm to 86,900 cfm and a hot air flow rate range of 20,200 cfm to 39,400 cfm. The optimized schedule causes a relatively larger flow range than the base schedule. However, this flow range increase can be accommodated by the existing system, which has a capacity of 90,000 cfm for cold air and of 50,000 cfm for hot air.

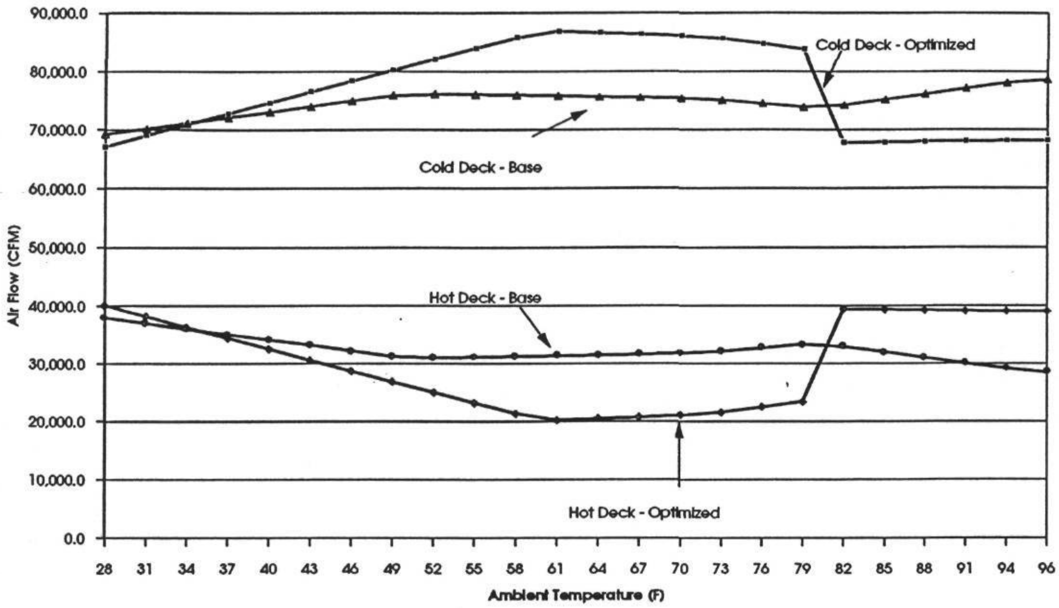

Figure 12: Comparison of Air Flow Rates through the Cold Deck and the Hot Deck under the Base and Optimized Schedules

The annual energy consumption has been predicted for both the base schedule and the optimized schedule and are compared in Figure 13. The horizontal axis is the ambient bin temperature and the vertical axis is the annual energy consumption for each bin year. The potential chilled water savings can be calculated as the areas enclosed by two chilled water consumption curves, and the potential Steam savings can be calculated as the area enclosed by two steam curves.

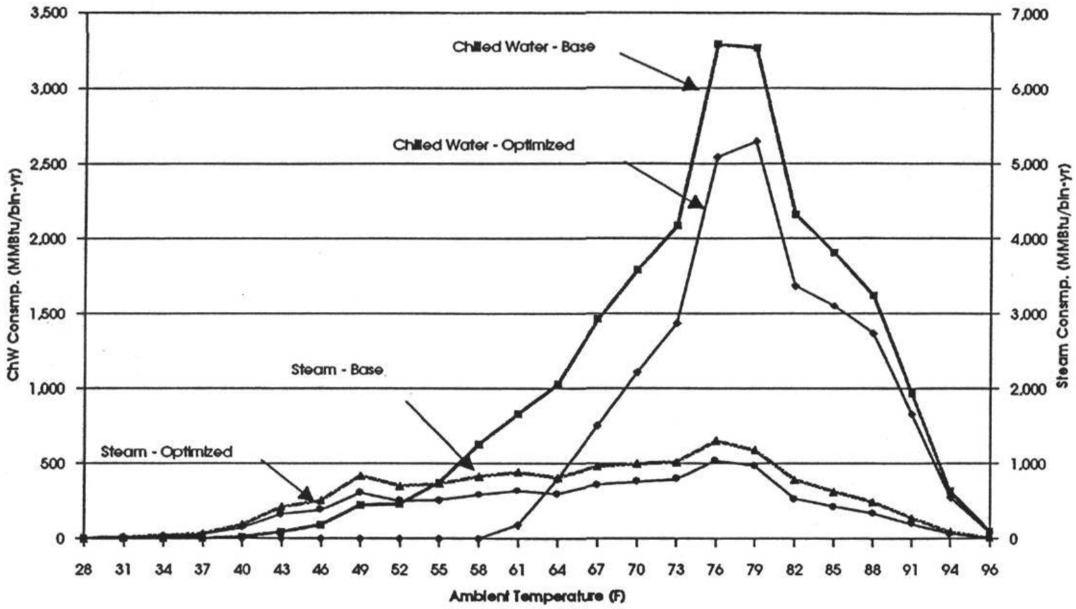

Figure 13: Comparison of the Predicted Annual Chilled Water and the Steam Energy Consumption under the Base and Optimized Operation Schedules

The overall energy performance and the potential savings are summarized in Table 5. It shows that the optimized schedule can reduce annual chilled water consumption from 22,300 MMBtu to 14,700 MMBtu, with a savings of 7,600 MMBtu/yr. and reduce the annual steam energy consumption from 13,700 MMBtu to 10,200 MMBtu with a savings of 3,500 MMBtu/yr. These energy savings reduce the annual cost by $55,700 for chilled water and $18,000 for steam. The total potential savings are $73,700/yr., which is 21% of the current annual building energy cost, or 32% of the current thermal energy costs.

Table 5: Summary of Potential O&M Savings at Clinical Science Building

Description	Consumption MMBtu		Savings MMBtu		Savings Dollars		Total	
	Ch-Water	Steam	Ch-Water	Steam	Ch-Water	Steam	Dollars	%
Base	22,300	13,700						
Optimized Case	14,700	10,200	7,600	3,500	55,700	18,000	73,700	21%

Note:

* The annual energy costs were $350,500, including $115,200 for electricity costs , $167,700 for chilled water costs, and $67,600 for steam (1992, Clinical Science Building, LoanSTAR measured energy consumption data).

* The energy costs were calculated according to the following unit energy prices: $0.02679/kWh for electricity, $7.30/MMBtu for chilled water and $5.055/MMBtu for steam.

Table 5 summarizes energy indices of Clinical Science Building based on gross floor area (124,870 ft^2). The optimized schedule can reduce annual chilled water consumption per unit floor area from 0.18 to 0.12 MMBtu/ft^2-yr., reduce steam energy index from 0.11 to 0.08 MMBtu/ft^2-yr. The potential chilled water and steam combination savings are \$0.59/ft^2-yr.

Table 5: Summary of Thermal Energy Indices

Item	Chilled water	Steam	Savings (MMBtu/ft^2-yr)		Savings
	MMBtu/ft^2-yr	MMBtu/ft^2-yr	Chilled water	Steam	(\$/ft^2-yr)
Base	0.18	0.11			
Optimized	0.12	0.08	0.06	0.03	0.59

6. CONCLUSIONS

Our study finds that the annual building energy costs can be reduced by \$73,600. The optimized operation schedules, developed by minimizing thermal energy consumption in the building, can be implemented by changing the EMCS program. The optimized operation schedule does not degrade the room comfort levels.

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ACKNOWLEDGMENTS

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APPENDIX A: SIMPLIFIED SYSTEM MODELS

The schematic of air handling unit (AHU) is shown in Figure A1. The building is idealized as two zones: interior zone and exterior zone.

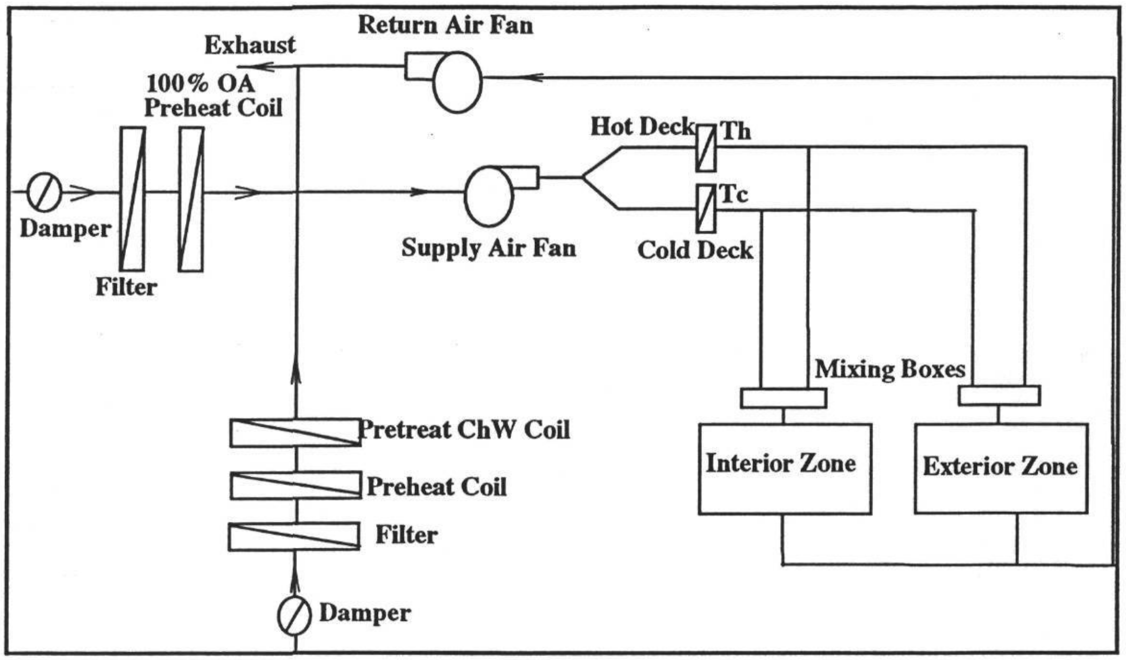

Figure A1: Schematic of HVAC System for Clinical Science Building

The chilled water consumption of the main cold deck is calculated by the formula:

$$E_c = \dot{m}_c (h_m - h_c)$$

where, E_c is the chilled water energy consumption of the main cold deck, $\dot{m}_c$ is the mass flow rate through the cold deck, h_m is the specific air enthalpy at the entrance of the cold deck and h_c is the cold deck supply air specific enthalpy.

The Steam energy consumption of the hot deck is calculated by the formula:

$$E_h = \dot{m}_h \times C_p (T_m - T_h)$$

where, E_h is the Steam energy consumption of hot deck, $\dot{m}_h$ is the mass air flow rate through the hot deck, T_m is the air temperature at the entrance of the hot deck, T_h is the hot deck supply air temperature and C_p is the air specific heat.

The pre-treatment cold deck is turned on only when ambient temperature is higher than 60 °F. The chilled water consumption due to this cold deck is calculated as:

$$E_{pre} = \dot{m} \times f_o (h_o - h_{pre}) + E_{fan-pre}$$

where E_{pre} is the chilled water consumption of pre-treatment cold deck, $\dot{m}$ is the total supply air mass flow rate, f_o is the outdoor air intake fraction, h_o is the outdoor air specific enthalpy, h_{pre} is the pre-treated supply air specific enthalpy, and $E_{fan\text{-}pre}$ is the energy consumed by the fan at the exit of the pre-treatment cold deck.

The preheat is turned on only when ambient temperature is lower than 55 °F. The Steam energy consumption is due to preheat is calculated as:

$$E_{preh} = \dot{m} \times f_o (h_{preh} - h_o)$$

where E_{preh} is the Steam consumption of preheat, $\dot{m}$ is the total supply air mass flow rate, f_o is the outdoor air intake fraction, h_o is the outdoor air specific enthalpy, h_{preh} is the preheat supply air specific enthalpy

The air specific enthalpy and temperature at the entrance of the cold deck and hot deck are calculated using energy balance principles.

$$h_m = f_o \times h_{pre} + (1 - f_o) \times h_r + \frac{E_{fan}}{\dot{m}}$$

where, h_r is the air specific enthalpy after the return air fan, E_{fan} is the energy consumption of the supply air fan and other symbols are as defined earlier.

The air temperature at the entrance of the cold deck and hot deck is also calculated using energy balance principles.

$$T_m = f_o \times T_{pre} + (1 - f_o) \times T_r + \frac{E_{fan}}{\dot{m} \times C_p}$$

where, T_{pre} is the pre-treatment cold deck supply air temperature, T_r is the return air temperature after the return fan, and other symbols are defined earlier.

The constant air flow terminal boxes are used in this building, therefore, the air flow rate through each box should not be changed. Consequently, the simplified model requires constant air flow rate to each zone although the ratio of cold air to the hot air changes with zone load and ambient, ambient condition and the cold deck and hot deck settings. The air flow rate to each zone is calculated according to the zone area.

$$\dot{m}_{ext} = \dot{m} \times \frac{A_{ext}}{A}$$
$$\dot{m}_{int} = \dot{m} \times \frac{A_{int}}{A}$$

where, $\dot{m}_{ext}$ and $\dot{m}_{int}$ are the air flow rate to exterior and interior zones respectively, A_{ext} and A_{int} are the conditioned floor areas in exterior and interior zones respectively and A is the total conditioned area.

The air flow through cold deck and hot deck can be solved through the following energy and mass balance equations:

$$\dot{m}_{c,int} \times (T_{room} - T_c) + \dot{m}_{h,int} \times (T_{room} - T_h) + \dot{m}_{inf,int} \times (T_{room} - T_o) = \frac{Q_{int}}{C_p}$$
$$\dot{m}_{c,ext} \times (T_{room} - T_c) + \dot{m}_{h,ext} \times (T_{room} - T_h) + \dot{m}_{inf,ext} \times (T_{room} - T_o) = \frac{Q_{ext}}{C_p}$$
$$\dot{m}_c = \dot{m}_{c,int} + \dot{m}_{c,ext}$$
$$\dot{m}_h = \dot{m}_{h,int} + \dot{m}_{h,ext}$$
$$\dot{m}_{ext} = \dot{m}_{c,ext} + \dot{m}_{h,ext}$$
$$\dot{m}_{int} = \dot{m}_{c,int} + \dot{m}_{h,int}$$

where, T_{room} is the room temperature, Q_{int} and Q_{ext} are the sensible loads at the interior zone and exterior zone respectively, $\dot{m}_{c,\,int}$ and $\dot{m}_{c,ext}$ are the cold deck air supply to the interior and exterior zones respectively, $\dot{m}_{h,int}$ and $\dot{m}_{h,ext}$ are the hot deck air supply to the interior and exterior zones respectively, $\dot{m}_c$ and $\dot{m}_h$ are the cold deck and hot deck air flow rate.

The room air specific humidity can be calculated using the following formula:

$$\omega_{int} = \frac{W_{int} + \dot{m}_{c,int} \times \omega_c + \dot{m}_{h,int} \times \omega_h + \dot{m}_{inf,int} \times \omega_o}{\dot{m}_{c,int} + \dot{m}_{h,int} + \dot{m}_{inf,int}}$$
$$\omega_{ext} = \frac{W_{ext} + \dot{m}_{c,int} \times \omega_c + \dot{m}_{h,ext} \times \omega_h + \dot{m}_{inf,ext} \times \omega_o}{\dot{m}_{c,ext} + \dot{m}_{h,ext} + \dot{m}_{inf,ext}}$$

where ω_{int} and $\omega_{ext,}$ are the room air specific humidity at the interior and exterior zones, respectively, W_{int} and W_{ext} are the moisture productions in the interior and exterior zones, respectively, ω_c and ω_h are the specific moisture levels at the exit of the cold deck and hot deck, respectively and other symbols are as defined earlier.

APPENDIX B: DATA QUALITY CHECK

The LoanSTAR measured chilled energy consumption is also compared with EMCS measured data for 24 hours from July 15, 1993 to July 16, 1993. Figure B1 shows the comparison results. The LoanSTAR measured chilled water consumption is with in 4% of measured data by EMCS.

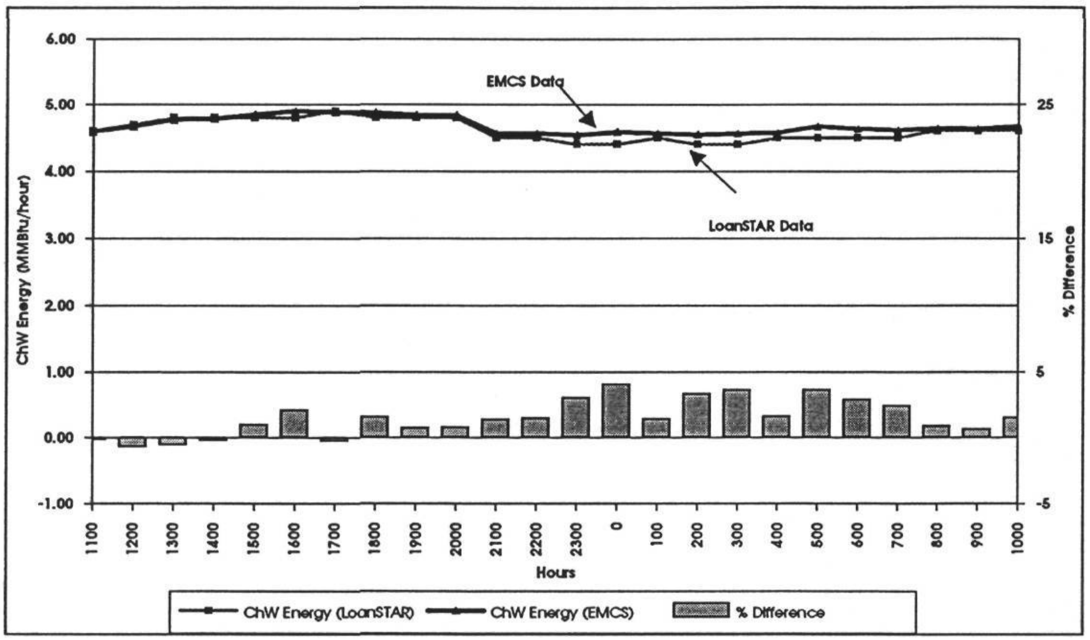

Figure B1: Comparison of LoanSTAR and EMCS measured hourly chilled water consumption data from July 15, 1993, 11:00 am to July 16, 1993, 10:00 pm.

Steam consumption data could not be compared with EMCS measured data due to the lack of EMCS data for the same period.